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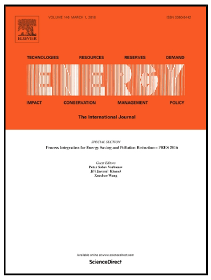

Analysis of Oil-free Linear Compressor Operated at High Pressure Ratios for Household Refrigeration

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Analysis of Oil-free Linear Compressor Operated at High Pressure Ratios for Household Refrigeration

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Abstract

Compared with conventional reciprocating compressor for vapour compression refrigeration (VCR) system, linear compressor offers higher energy efficiency and oil-free operation, which allows the use of mini/micro-channel heat exchangers. However, there are key challenges when oil-free linear compressors are used for household refrigeration with typical high pressure ratios (above 10), such as high clearance loss, high piston offset, and very non-linear gas spring. Previous papers by the author have demonstrated the feasibility of oil-free linear compressor for electronics cooling at lower pressure ratios (below 3.5). This paper presented comprehensive analysis of these issues as a key step towards developing oil-free linear compressor for household refrigeration. The model of non-linear gas spring at high pressure ratios is validated by measurements of a previous prototype linear compressor with minimum flow. Piston offset can be effectively controlled by solenoid valve at 1 Hz. Gas leakage increases by a factor of 2.5 if the piston is fully eccentric in the cylinder. The gas leakage loss can be 27% of power input for pressure ratio of 13.6 using R600a.

Keywords: linear compressor, household refrigeration, resonant frequency, gas spring, gas leakage, high pressure ratio

NOMENCLATURE

A	area (mm^2)
BLDC	brushless direct current
c	radial clearance (μm)
D	piston diameter (mm)
F	force (N)
f	frequency (Hz)
k	spring stiffness (N/mm)
L	length (mm)
LVDT	linear variable differential transformer
m	mass (g)
P	pressure (bar)
PID	proportion-integration-derivative
PWM	pulse width modulation
R	piston radius (mm)
S	stroke (mm)
T	temperature (°C)
V	volume (mm^3)
W	work (joule)
x	displacement (mm)
μ	viscosity (Pa·s)
ρ	density (kg/mm^3)
δ	eccentric distance (mm)
θ	angle (deg)
ε	distance between piston and cylinder wall (mm)
β	damping coefficient (N·s/m)

1. Introduction

Vapour compression refrigeration (VCR) system consists of refrigerant, compressor, condenser, expansion device and evaporator. VCR system has been widely used in domestic refrigerators and air conditioning of buildings and automobiles since its invention in early 20th century. Energy consumption increases exponentially with the rise of the numbers of refrigerators and air conditioners worldwide. Currently, there are about 1 billion units of refrigerators worldwide **[1]**. It is acknowledged that refrigeration and air conditioning systems are responsible for roughly 30% of total energy consumption, therefore unquestionably with a major impact on energy demand **[2]**. This also causes a dramatic increase in the emissions released from these VCR systems, which has a significant, negative environmental impact. The development of VCR system towards high energy efficiency, low emissions and small size requires new technologies for each component. Linear compressor appears to over perform conventional crank-driven compressor **[3]**. Compact heat exchangers such as mini/micro-channel reduce the charge of refrigerant in the VCR system thus the emissions that cause the global warming **[4]**.

Linear compressors do not have crank mechanism for reciprocating and is driven directly by a linear motor. Frictional loss is reduced significantly. Oil-free operation is possible, and this is a significant advantage when compact heat exchangers are used in a VCR system. The absence of oil widens both the choice of refrigerants and their operating temperature range. Eliminating the need for the oil return reduces constraints on pipe sizing and will lead to a reduction in pressure drop losses **[5]**. Linear compressor is capable of operating at resonant frequency to minimise the input current. This will reduce the energy consumption further. The potential for this technology to be scaled to small physical sizes is better than for conventional compressors **[6]**. The refrigeration capacity for system using linear compressors can be modulated by simply changing the excitation voltage. It can be seen that linear compressors are attractive for applications in domestic refrigeration compared with crank-driven compressors. Linear compressor typically consists of piston/cylinder assembly, linear motor and suspension spring as shown in Fig. 1. Note that both flexure and coil springs can be used for the suspension to align the piston with the cylinder. Radial clearance is in the range of 10-20 μm for oil-free design and much smaller when lubricant oil is used.

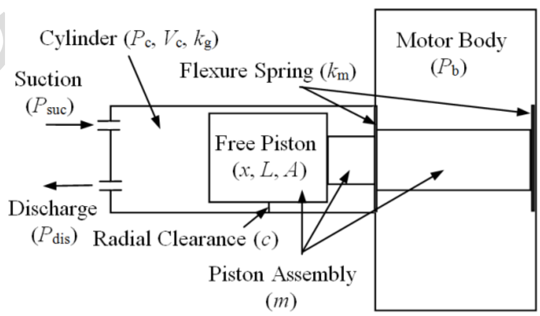
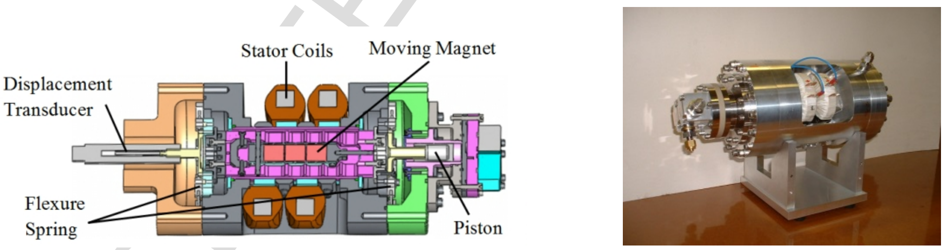


Fig. 1 Schematic of the linear compressor model showing key parameters (adapted from Davies et al. **[7]**)

With many years of successful application in space cryocooler where no oil lubricant can be tolerated, linear compressors are recently being extended for domestic market. Korean company LG has marketed linear compressor for household refrigeration since 2002 **[8]**, with technology licensed by Sunpower **[9]**. Despite Sunpower's early machines being oil free, the LG compressors designed for R134a and R600a are not oil free and they do not have the gas bearing system. In 2014, Brazil company Embraco launched an oil-free linear compressor for household refrigeration **[10]**. There is no published data and it appears that there is no large scale manufacture of this design. A small moving magnet linear compressor with a motor efficiency of 71% designed by Embraco was utilized by Possamai et al. **[11]** for developing a prototype VCR system using micro-channel heat exchangers for a laptop cooling application using R600a. Bijanzad et al. **[12]** built an oil-free solenoid based linear compressor for household refrigerator. Results show that 85% motor efficiency was achieved at 19 Hz. The major technical issues for linear compressor in a household refrigeration system at high pressure ratios include high power loss due to the gas leakage across the radial clearance, high piston offset/drift due to the pressure difference between the mean cylinder pressure and the body pressure [13, and 14], and very non-linear gas spring stiffness [15 and 16]. This paper presented latest work on the analysis of oil-free linear compressor operated at higher pressure ratios for domestic refrigeration. Key issues listed above and possible solutions are addressed in this paper. The prototype oil-free linear compressor was designed for cooling electronics at lower pressure ratios (below 3.0 using R134a). Fig. 2 shows the oil-free linear compressor, consisting of a moving magnet linear motor, flexure spring and virtual radial clearance of about 10 µm. The modelling and measurements of this oil-free linear compressor using R134a with lower pressure ratios operation have been published in **[17 and 18]**. This paper will address the three key challenges by further measurements of the original prototype (designed for low pressure ratios) and modelling of a proposed design for high pressure ratios.



(a) (b)

Fig. 2 Oil-free linear compressor: (a) moving magnet linear compressor configuration (Liang et al. **[15]**); (b) prototype linear compressor for lower pressure ratios

2. Instrumentation on the Prototype Linear Compressor

The prototype linear compressor has a maximum stroke of 14 mm, piston diameter of 19 mm, and maximum shaft power output of 100 W. Two identical compressors operate opposite to each other with shared discharge and suction lines. The pressure ratio for the design point is 3.0 for VCR system that is used for electronics cooling. The linear compressor has been operated at high pressure ratios up to 6.9 with minimum flow for this work. The flexure spring has a stiffness of 17 kN/m. Liang et al. **[15]** reported details of the test rig for linear compressor. A series of tests have been conducted by using nitrogen **[18]** and refrigerant R134a **[17]** at pressure ratios below 3.5. Fig. 3 shows the the test rig. The displacement of the piston was measured by using a linear variable differential transformer (LVDT) displacement transducer. One mass flow meter, four pressure transducers and eight thermocouples were used. A LabVIEW programme logged these parameters and controlled the voltage to the compressor thus the piston stroke. Performance of the prototype linear compressor such as motor efficiency, resonant frequency, pressure-volume (P-V) loop at lower pressure ratios (below 3.5) has been reported by Liang et al. **[17 and 18]**.

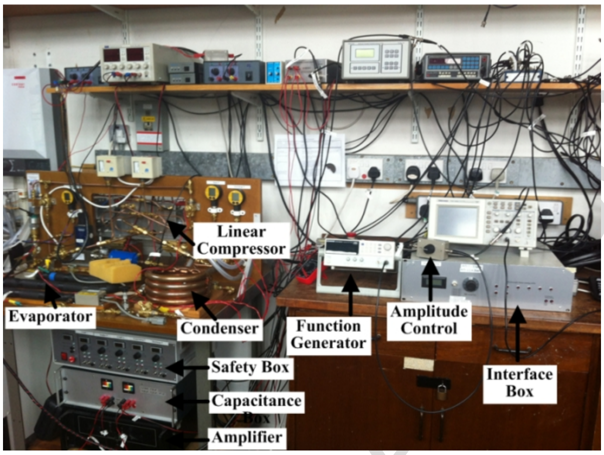


Fig. 3 Instrumentation on the prototype linear compressor designed for electronics cooling

In order to operate the prototype linear compressor at higher pressure ratios (over 3.5), a minimum flow of nitrogen was used. Although this is not the target operating point for this prototype, the experimental data can be used to study the very non-linear gas spring behaviour at high pressure ratios. This will enable a proposed design of linear compressor for household refrigeration at high pressure ratios. For this purpose, a series of tests have been conducted for pressure ratios between 3.5 and 6.9. The in-cylinder pressure P_c was not measured by inferred from the dynamics of the piston (shown in Fig. 1) as follow:

$$P_c = (F_s - m\ddot{x} - k_m x - \beta\dot{x})/A + P_b \tag{7}$$

where F_s is the shaft force from the linear motor, m is the moving mass, x is the displacement of the piston, k_m is the stiffness of the mechanical flexure spring, β is the damping coefficient, A is the piston area, and P_b is the body pressure.

Resonant frequency for each pressure ratio has been manually identified during the compressor operation by looking for minimum current input for each piston stroke.

3. Power Loss and Compressor Efficiency

3.1 Power Loss due to Gas Leakage

Although oil-free operation demonstrates a lot of benefits, the absence of oil will increase the gas leakage across the radial clearance (10-20 μm) which causes a power loss. The leakage can be described as a steady flow through an annulus, as shown in Fig. 4. The mean mass flow rate across the radial clearance can be described as follow:

$$\dot{m}_1 = \frac{\pi D c^3 (P_1 + P_2)}{12\mu L \quad 2R_g T_0}(P_1 - P_2) \tag{1}$$

where c is the radial clearance, D is the piston diameter, μ is viscosity, L is the length of clearance, P_1 is the inlet pressure and P_2 is the pressure on the other side.

The power loss due to the gas leakage becomes

$$\dot{W}_1 \approx \frac{\pi D c^3}{24\mu L}\left(P_1^2 - P_2^2\right)\left[\frac{(P_1 - P_2)}{P_1}\right] \tag{2}$$

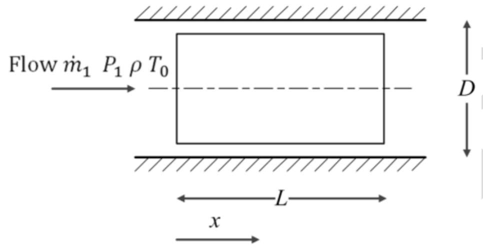


Fig. 4 Flow through a clearance seal with radial clearance c

It can be seen that higher pressure ratio (P_1/P_2) will significantly increase the power loss due to gas leakage. Another key issue is even higher loss occurs when piston and cylinder become eccentric due to manufacture tolerance and side electromagnetic force. The leakage across a concentric clearance can be simplied as

$$\dot{m}_1 = \frac{k\pi D c^3}{L} \tag{3}$$

where k is the coefficient for a certain pressure drop.

Fig. 5 gives the diagram of full eccentricty of piston in the cylinder. From Equation (3), with a constant value of piston diameter, overlap length and pressure across the seal, the leakage flow for a concentric piston will be

$$\dot{m}_1 = k \cdot 2\pi R\delta^3/L \tag{4}$$

In the case of eccentricty, the leakage flow is

$$\dot{m}_1 = k * 2\int_0^{\pi}\varepsilon^3 R d\theta/L \tag{5}$$

As $\varepsilon \approx \delta(1 + cos\theta)$, the flow becomes

$$\dot{m}_1 = k * 5\pi R\delta^3/L \tag{6}$$

It can be seen that gas leakage inreases by a factor of 2.5 if the piston is fully eccentric in the cylinder. The leakage will also be influenced by any tilting of the piston and any tapering of either the piston or the cylinder. As can be seen from the Equation (6), the radial clearance has much more influence on the leakage than the length of clearance.

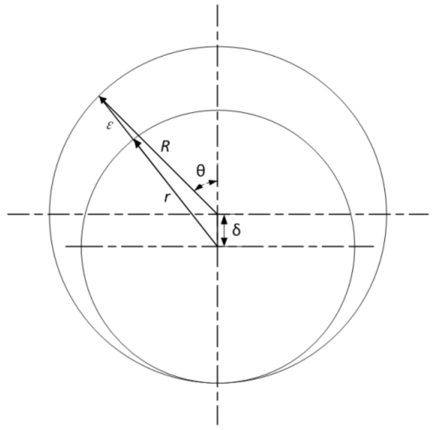


Fig. 5 Schematic for leakage when piston is fully eccentric

3.2 Compressor Efficiency

Fig. 6 shows the motor efficiency, power factor and compressor stroke for minimum flow measurement at high pressure ratios from 3.5 to 6.9 using test rig shown in Fig. 3. The compressor stroke is 13.6 mm for pressure ratio of 6.9. Motor efficiency is above 94% and power factor is above 0.96, indicating that the linear compressor was operated at its resonance for each pressure ratio.

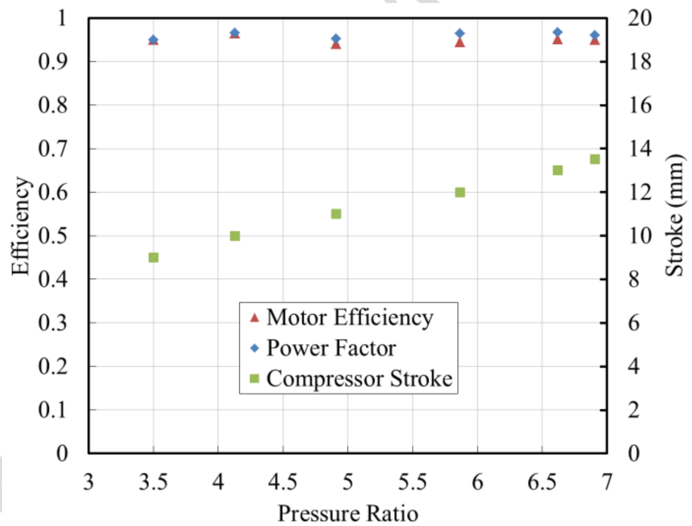


Fig. 6 Efficiency, power factor and stroke of the linear compressor at high pressure ratios

Fig. 7 shows the power loss due to gas leakage according to Equation (2) and the power input into the linear compressor. A radial clearance of 10 μm was assumed. It can be seen that seal leakage loss increases faster than power input as pressure ratio increases. At a pressure ratio of 6.9, the seal leakage loss represents 8.2% of the power input. This indicates that the overall efficiency of the linear compressor will not be significantly higher than conventional compressor at high pressure ratios.

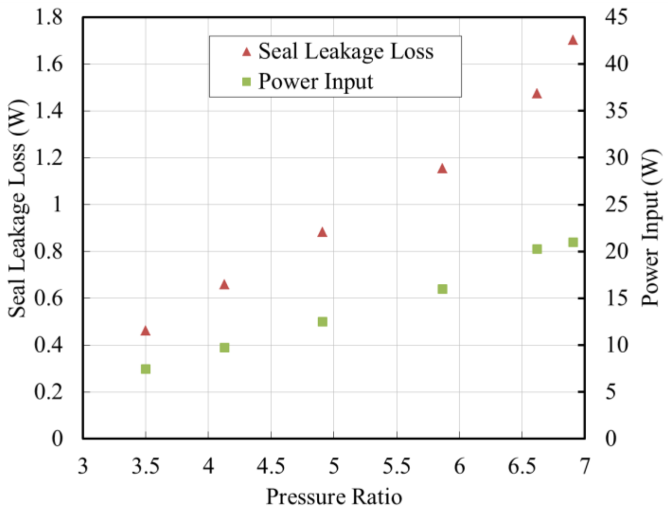


Fig. 7 Seal leakage loss and power input of the linear compressor at higher pressure ratios

4. Non-linear Gas Spring and Resonant Frequency

4.1 Non-linear Gas Spring

When operated at resonant frequency, the current input can be minimised so that the motor efficiency can be higher. The resonant frequency varies with operating conditions. The resonant frequency of the linear compressor f is determined by the mechanical spring and gas spring as follow:

$$f = \frac{1}{2\pi}\sqrt{\frac{k_g + k_m}{m}} \tag{8}$$

where k_g is the stiffness of gas spring.

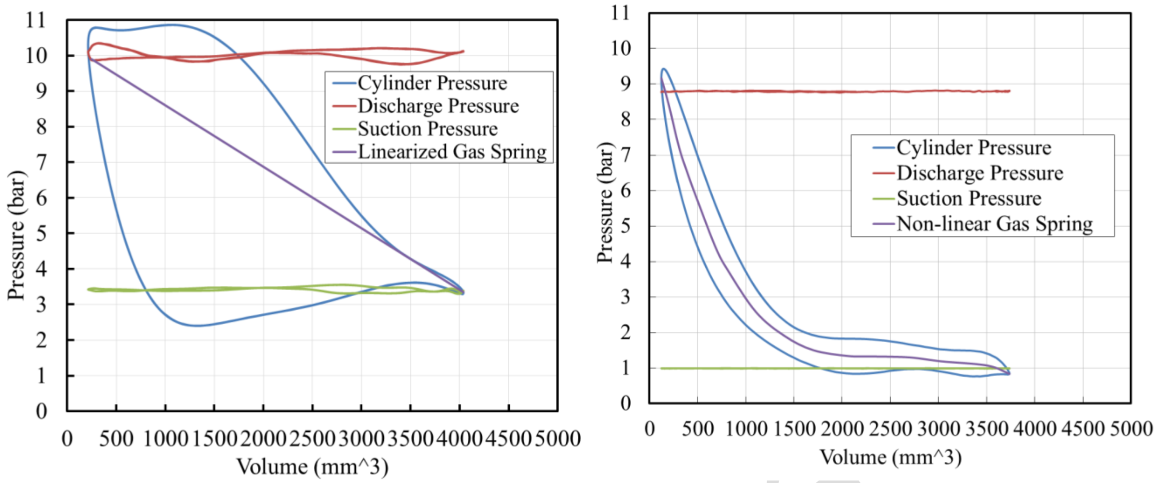
For low pressure ratios (below 3.5), the gas spring can be estimated by linearizing the gas force over the compression and discharge processes as shown in Fig. 8.

Liang et al. **[17]** validated this simplified resonance model by measuring the resonant frequency for each operating condition. However, when the linear compressor is operated at high pressure ratios, since the gas spring is highly nonlinear and variable throughout the operation, the linearized resonance model is not accurate. Although this can be solved by using a much stiffer mechanical spring as has been proposed by LG **[8]**, the size, weight and cost of the linear compressor will all increase. Prediction of resonant frequency at higher pressure ratios is necessary.

The very non-linear gas spring curve (purple curve in Fig. 8b at a pressure ratio of 9.0) can be calculated by averaging the cylinder pressure at compression and expansion. The energy stored in the gas spring can be expressed as

$$\frac{k_g(S)^2}{2} = \int_1^2 P_c dV - W_s/2 \tag{9}$$

where 1 is the start of compression and 2 is the end of discharge. The shaft work W_s can be calculated as the enclosed area of the *P*-*V* loop.



(a) (b)

Fig. 8 *P*-*V* loops with linear gas spring at pressure ratios of 2.5 (a) and 9.0 (b) using nitrogen

4.2 Resonant Frequency

Fig. 9 compares the modelled resonant frequency using non-linear model with measured values for high pressure ratios. The predictive model agrees well with measurement, with an average error of 2%. This indicates that the non-linear gas spring can still be accurately predicted for each operating condition so that the linear compressor can always operate at resonance.

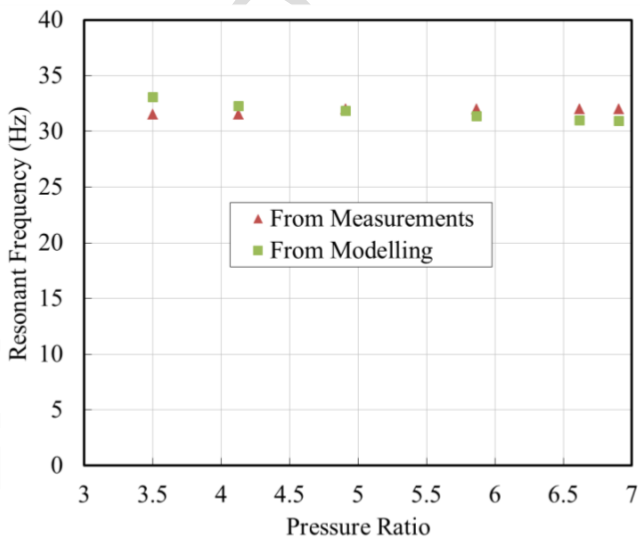


Fig. 9 Predicted resonant frequency against measurements at high pressure ratios

5. Piston Offset

5.1 Piston Offset Modelling

The problem of piston offset is caused by the differential pressure generated across a clearance seal which has a fluctuating pressure on one side of it (piston-cylinder) and a constant pressure on the other (body). On the body side of the seal, the pressure is essentially constant, whereas the working side of the seal experiences a fluctuating pressure. The pressure differential is given by

$$\Delta P = P_{\mathrm{c,\ m}} + P_{\mathrm{c,a}}\sin(2\pi f t) - P_{\mathrm{b}} \tag{10}$$

where $P_{\mathrm{c,\ m}}$ is the mean in-cylinder pressure and $P_{\mathrm{c,a}}$ is the amplitude of the in-cylinder pressure.

If the body pressure P_{b} is equal to the mean in-cylinder pressure $P_{\mathrm{c,\ m}}$, then the net volumetric flow taken round one cycle is zero. The mass flow rate, however, is proportional to the mean density, and this is higher when the piston is closer to cylinder head than when it is closer to compressor body side. Thus there will be a net mass flow from the working side to body volumes across the radial clearance seal, which will decrease the mean in-cylinder pressure and increase the body pressure. Eventually an equilibrium point will be reached when this effect is counterbalanced by the pressure differential in the opposite direction.

If there is a difference between the mean in-cylinder pressure and the body pressure there will be a net axial force which is counteracted by the mechanical springs, and will result in a shift of the mean position of the piston. Note that if the piston is not oscillating about the 'mechanical zero' of the springs, there will be a reduction in the useful stroke of the compressor. The piston offset x_0 can be calculated as follow:

$$x_0 = \frac{(P_{\mathrm{c,\ m}} - P_{\mathrm{b,m}})\mathrm{A}}{k_{\mathrm{m}}} \tag{11}$$

where in-cylinder pressure $P_{\mathrm{c,\ m}}$ can be calculated from the *P-V* loop for each operating condition.

Fig. 10 shows the piston displacement from measurement at a stroke of 10 mm at a pressure ratio of 5.0. The piston offset is -1.45 mm. Note negative displacement means towards the cylinder head. This means the piston oscillates about a position away from the datum position. The maximum stroke will be much lower than the value that the linear compressor was designed for. Body pressure varies due the gas leakage across the radial clearance. The mean body pressure is 5.97 bar.

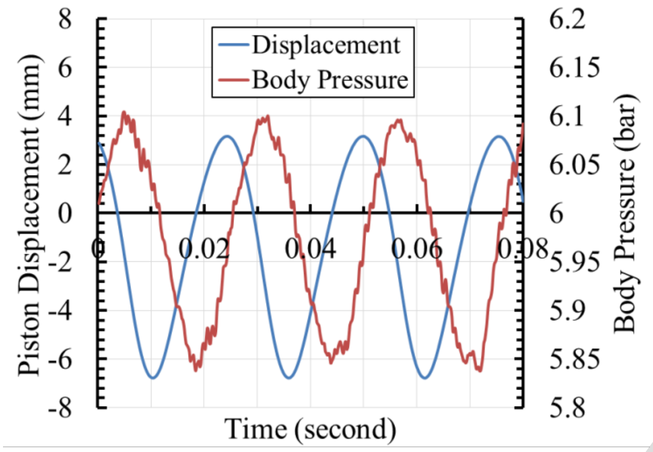


Fig. 10 Piston displacement and variation of body pressure from measurements

Fig. 11 shows the piston offset against stroke at a pressure ratio of 5.0 from both measurements and calculations when there is no control on the offset. Calculations agree well with measurements indicating the accuracy of the calculating in-cylinder pressure as well (see Equation 7). As the pressure ratio increases, the piston offset will further increase. It can be seen that control is required particularly for higher stroke because high piston offset will lead to collision between the piston and cylinder head.

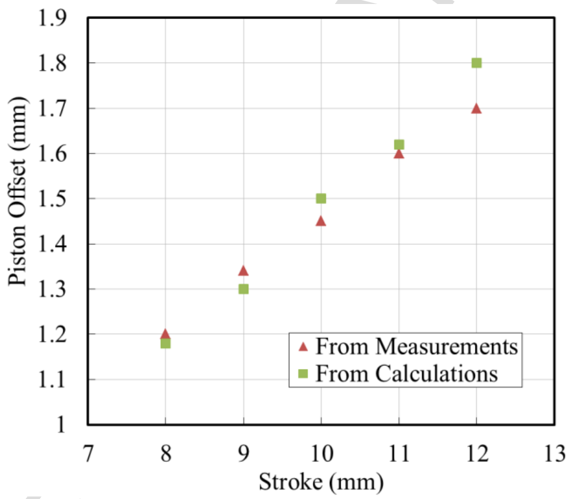


Fig. 11 Piston offset against stroke at a pressure ratio of 5.0

5.2 Piston Offset Control

A bleed flow connecting the body pressure back to the suction line has been added to the main flow loop. The bleed flow was controlled by solenoid valve (adapted from a fuel injector of a gasoline engine). A PID (proportion-integration-derivative) controller has been developed to modulate the pulse width of the solenoid in order to keep the piston offset at zero. Fig. 12 shows the bleed flow loop using solenoid valve with flow measured by a mass flow meter. A needle valve was constructed before the use of PWM (pulse width modulation) for manual adjustment of piston offset.

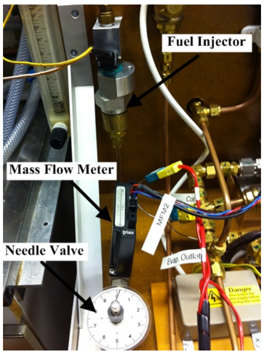


Fig. 12 Bleed flow loop for piston offset control using solenoid valve adapted from fuel injector

Although solenoid valve can effectively control the piston offset, it also increase extra cost for mass production. High frequency operation of the solenoid valve could cause reliability issue and thus require maintenance. A series of pulses with a frequency of 40 Hz and changing duty cycle were generated to reduce the piston offset to 0 mm after 50 s. A series of control system tests have been conducted by changing the PWM frequency from 1 to 40 Hz. The settling time remains the same for different PWM frequencies when the PID gains are fixed. The duty cycle varies from 4.4% at 10 Hz of PWM to 6.8% at 1 Hz. A 12 V DC power supply was used to drive the solenoid with a resistance of 12 Ω. With a steady-state duty cycle of 6.8%, the electrical power consumption is 0.82W, which is negligible. A 1 Hz solenoid will significantly increase the durability and reduce the cost as well.

6. Predictive Performance using R600a

Analysis on the gas leakage, non-linear gas spring and piston offset shows that oil-free linear compressor is capable of low cost control and high efficiency at higher pressure ratios. However, the prototype linear compressor mentioned above was not designed for household refrigeration at high pressure ratios. A compressor model for household refrigeration has been built to investigate the performance of oil-free linear compressor at higher pressure ratios using R600a (iso-butane) as refrigerant. A commercial crank-driven compressor has been used as reference and data source. It is assumed that reed valve and heat transfer through the cylinder wall are same for both linear compressor and crank-driven compressor. The model will provide baseline performance data for the prototyping of oil-free linear compressor for the household refrigeration.

Fig. 13 shows the *P-V* loop from a crank-driven compressor at a pressure ratio of 13.6 using R600a. The evaporator and condenser temperatures are -23.3 °C and 60 °C respectively. The polytropic index for the compression stroke is calculated to be 0.93 and 0.80 for the expansion stroke. For R600a at -23.3/+60°C, the saturation pressures are 0.63 and 8.7 bar respectively. The cylinder bore is 25.4 mm and the compressor stroke is 22.8 mm. The shaft power of the crank-driven compressor was calculated to be 72 W which indicates a shaft work of 1.44 J at ASHRAE condition (Standard 23.1-2010). Green curve in Fig. 13 is reconstructed *P-V* loop by assuming similar polytropic index for compressor and expansion.

The mean gas pressure (red curve) has been calculated according to approach in Section 4 for predicting the resonant frequency.

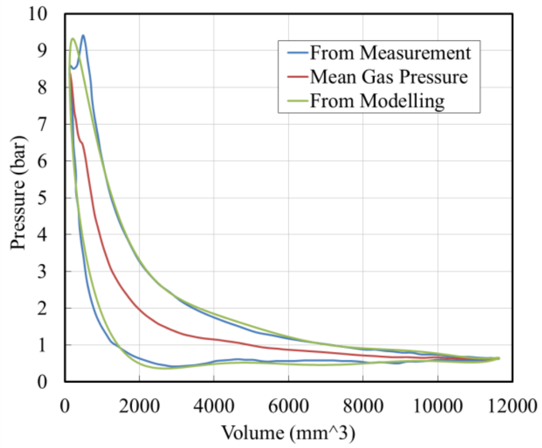


Fig. 13 *P-V* loop for crank-driven compressor and reconstructed loop with mean gas pressure at pressure ratio of 13.6 using R600a

By using the simplified compressor model, mean cylinder pressure, effective gas spring stiffness and shaft power can be calculated for comparison. Table 1 compares the model and test. Good agreements have been achieved. This simplified compressor model can then be used to predict performance of a new linear compressor with comparable parameters to crank-driven compressor for household refrigeration.

Table 1 key result from two *P-V* loops for a pressure ratio of 13.6

	Compressor model	Compressor test
Mean cylinder pressure (bar)	1.99	1.92
Gas spring stiffness (N/mm)	18.6	18.5
Shaft power (W)	69.5	72

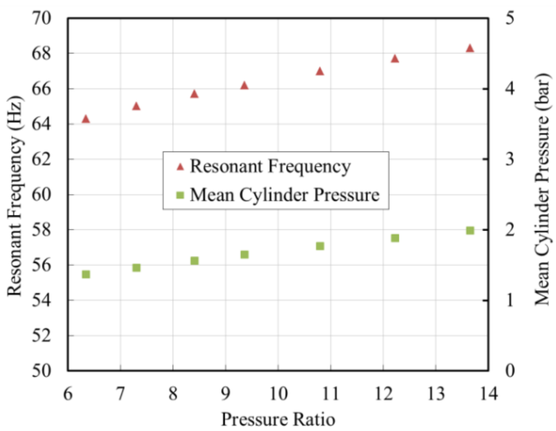
Table 2 gives the design parameters for a linear compressor for household refrigeration. The predicted shaft power is 31 W so that two compressors opposite to each other will be required. It can be seen that effective gas spring stiffness is much higher than mechanical spring stiffness. This means that resonant frequency will vary with different ASHRAE conditions.

Table 2 Linear compressor design parameters for R600a (two halves)

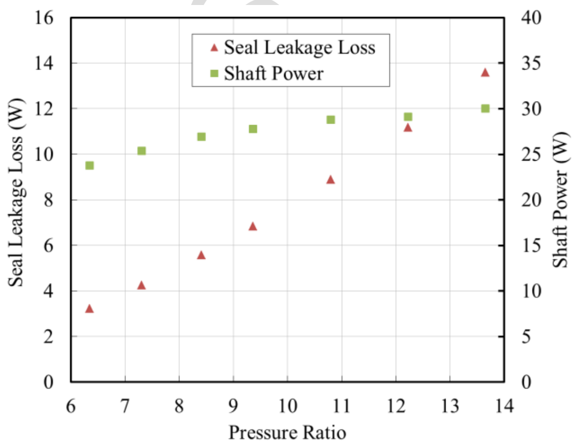
Maximum stroke (mm)	8
Maximum shaft force (N)	100
Moving mass (kg)	0.15
Mechanical spring stiffness (N/mm)	10
Piston diameter (mm)	20
Dead volume (mm^3) at max stroke	100
Refrigerant	R600a

Radial clearance (μm)	10

Fig. 14 shows predicted performance of the proposed linear compressor for household refrigeration. Evaporator temperature is fixed at -23°C and the condenser temperature varies from 40°C to 60°C. Mean cylinder pressure increases from 1.6 bar to 2.0 bar for pressure ratios of 6.4-13.7. This means that the piston offset could be significantly high. Resonant frequency varies from 64 Hz to 69 Hz for the range of pressure ratios. When the condenser temperature is fixed at 60°C, the resonant frequencies for evaporator temperatures of -10°C, -23°C and -30°C are 79 Hz, 68 Hz, and 64 Hz respectively. This indicates the importance of instantaneous track of resonant frequency for the linear compressor. Seal leakage loss increases sharply with pressure ratios. Assume a motor efficiency of 90%, the seal leakage loss represents 27% of the total input power. This is a significant loss of the linear compressor when there is no oil lubricant.



(a)



(b)

Fig. 14 Predicted performance of a linear compressor for household refrigeration using R600a: (a) resonant frequency and mean cylinder pressure; (b) seal leakage loss and shaft power

7. Conclusions

Analysis of an oil-free linear compressor at higher pressure ratios has been conducted in this work. Both modelling and measurements have been reported in this paper. Key findings are as below:

(1) Gas leakage inreases by a factor of 2.5 if the piston is fully eccentric in the cylinder. At a pressure ratio of 6.9 using nitrogen, the seal leakage loss represents 8.2% of the power input. This is significant power loss of linear compressor.(2) The gas spring becomes very non-linear at higher pressure ratios. The proposed approach to calculating the effective gas spring stiffness has been validated by operating the linear compressor at higher pressure ratios with minimum flow of nitrogen as working fluid.

(3) A bleed flow using 1 Hz solenoid valve is capable of controlling the piston offset efficiently with much lower cost and high durability. The duty cycle of 6.8% at 1 Hz causes

an additional electrical power consumption of 0.82 W, which is negligible in comparison with the total electrical power into the linear compressor.

(4) For a comparable linear compressor design using R600a, when the condenser temperature is fixed at 60°C, the resonant frequencies for evaporator temperatures of -10°C, -23°C and -30°C are 79 Hz, 68 Hz, and 64 Hz respectively. The seal leakage loss can be 27% of power input. This is a key reason for using oil lubricant to prevent such loss.

Acknowledgement

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- Gas leakage inreases by a factor of 2.5 if the piston is fully eccentric in the cylinder.
- The approach to estimating nonlinear gas spring stiffness has been validated.
- A 1 Hz solenoid valve can be used for piston offset control with 0.82 W power input.
- The seal leakage loss is 27% of power input for pressure ratio of 13.6 using R600a.